UNCLASSIFIED COPY NO. 6

U·S·NAVAL·TECHNICAL·MISSION·IN·EUROPE

ATI No. 36156

CADO FILE COPY

~~CONFIDENTIAL~~

RESTRICTED

TECHNICAL REPORT NO. 474-45

AD A953135

MAIN PROPULSION SHAFTING AND BEARINGS OF GERMAN NAVAL VESSELS

September 1945

DTIC FILE COPY

E

U·S·NAVAL·TECHNICAL·MISSION·IN·EUROPE

UNCLASSIFIED

DISCLAIMER NOTICE

THIS DOCUMENT IS BEST QUALITY PRACTICABLE. THE COPY FURNISHED TO DTIC CONTAINED A SIGNIFICANT NUMBER OF PAGES WHICH DO NOT REPRODUCE LEGIBLY.

U. S. NAVAL TECHNICAL MISSION IN EUROPE
c/o Fleet Post Office
New York, N.Y.

File: A9-16(3)(10/Hn)

Serial: 01275

10 October 1945

~~CONFIDENTIAL~~

From: Chief, U.S. Naval Technical Mission in Europe.
To : Chief of Naval Operations (OP-16-PT).

Subject: U.S. Naval Technical Mission in Europe Technical Report No. 474-45, Main Propulsion Shafting and Bearings of German Naval Vessels - Forwarding of.

Enclosures: (A) (HW) Four (4) complete copies of subject report as listed in distribution.
(B) (HW) One (1) set of negatives of photographs in subject report with copy No. 5 as listed in distribution.
(C) (HW) Original German Document of the Scope of Vibration by Dr. R. Hoppernath, with copy No. 5 as listed in distribution.

1. Enclosures (A) (B) and (C) are forwarded herewith.

2. CNO (OP-16-PT) is requested to make complete additional copies of this report for forwarding to such other agencies as may be interested.

Harry D. Hoffman
Acting

DISTRIBUTION To:

CNO (OP-16-PT)	Copy Nos. 1 - 4
CNO (OP-16-PT) ...... w/Enclosures B and C	Copy No. 5
FIAT	Copy No. 6
Office of Research and Invention	Copy No. 7
ComNavEu	Copy No. 8
ComNavEu for Admiralty	Copy No. 9
Naval Research Laboratory	Copy No. 10
BuShips	Copy Nos. 11 - 12
BuShips (Code 332)	Copy No. 13
BuShips (Code 443)	Copy No. 14
BuShips (Code 634)	Copy No. 15
David Taylor Model Basin, Carderock, Md.	Copy No. 16
NavTecMisEu (Main Files)	Copy No. 17

[redacted]

TECHNICAL REPORT NO 474-45.

MAIN PROPULSION SHAFTING AND BEARINGS OF GERMAN NAVAL VESSELS

SUMMARY

This report covers the design calculations and constructional details of the propulsion shafting of a German cruiser of 38750 S.H.P. per shaft. Additional data regarding bearings is also included.

September 1945

U. S. NAVAL TECHNICAL MISSION IN EUROPE

CONFIDENTIAL

TABLE OF CONTENTS

		Page
I.	Conclusions.	4
II.	Plants Visited and Engineers Contacted.	5
III.	Design Details.	
	(1) Shafting.	5
	(a) Materials.	5
	(b) Corrosion Protection.	5
	(c) Shaft Flanges.	6
	(d) Synchronizing Couplings.	6
	(2) Bearings.	
	(a) Outboard.	6
	(b) Line Shaft.	7
IV.	Design Calculations.	
	(1) General Data.	7
	(2) Calculations for Outboard Propeller Shaft, Section I - I, Fig. 2.	7
	(3) Calculations for Outboard Propeller Shaft, Section II - II, Fig. 2.	9
	(4) Calculations for Outboard Line Shaft.	10
	(5) Calculations for Propeller shaft coupling.	11
	(6) Calculations for Flange Bolts.	12
	(7) Calculations for Key for Outboard coupling.	13
	(8) Calculations for Middle Shaft.	13
	(9) Strength of Middle Shaft with Astern Operation.	13
	(10) Strength of a Locked Outboard Shaft at a Speed of 14 knots.	14
	(11) Safety of Shafting Against Buckling.	16
V.	References.	
	(a) FESTIGKEITSBERECHNUNG DER WELLENLEITUNG (Strength Calculations for Shafting) Photostat forwarded to CNO(OP-16-PT).	

CONFIDENTIAL

TABLE OF CONTENTS (Cont'd).

References (Cont'd).
(b) NavTecMisEu Technical Report No. 138-45
pp 28-32 and plate T-58.

Appendix

Translation of the Scope of Vibration Work done by Dr. R. Hoppenrath, of Krupp Germania Werft, Kiel.

CONFIDENTIAL

MAIN PROPULSION SHAFTING AND BEARINGS

OF GERMAN NAVAL VESSELS

I. Conclusions.

(1) The method of calculating stresses in main propulsion shafting of German geared turbine driven Naval vessels is essentially the same as was used by the Bureau of Ships up to about 1940. It follows closely the German or British Lloyd methods. In this type of drive, torsional vibration is excited only by the propeller, and stresses due to this cause are so low that they can be neglected. Factors of safety used by the German shipbuilders, based upon ultimate tensile strength of the material, are of the same order of magnitude as those which were used by the Bureau of Ships.

(2) Shaft calculations involving the determination of vibratory stresses and forced frequencies, such as are encountered in Diesel driven ships, are made by a specialist, in the case of German shipyards. The calculations are made according to the method of Holzer or Tolle and the effect of stress concentrations is taken into account. In general, a stress concentration factor of two (2) is used at a point of discontinuity. A similar procedure is used by the Bureau of Ships, American shipyards, and manufacturers.

(3) Measurements of frequencies and vibratory amplitudes are made of the mass-elastic system of all new installations. A description of the instruments used for this purpose is given in a report by another member of the NavTecMisEu.

(4) Conversations with Blohm and Voss engineers brought out the fact that this company had built, during the last year, propeller strut bearings which were much shorter than previously accepted standard designs. The short bearings were found to operate more satisfactorily than the longer ones. This fact is of interest to the Navy, especially since a similar proposal has been made in the Bureau of Ships.

(5) The German Navy made large use of disc lubricated line shaft bearings. These have been proposed in lieu of ring lubricated bearings now exclusively used by the Navy. No difficulties were reported with the disc lubricated bearings used by the Germans.

(6) Bearing loads for shaft keys used by German shipyards are somewhat higher than used by the U.S. Navy. The former uses approximately 29000 PSI while the Navy specifies a maximum of

Conclusions(6), (Cont'd).

25000 PSI.

(7) German shipyards do not make a full scale calibration test of the section of shafting on which a torsion meter is to be used. The subject of torsion meters is being covered in a separate report being prepared by another member of the NavTecMisEu.

(8) It will be of interest to the Navy to compare the German method of computing torque and pull developed by a locked propeller with the method developed by the David Taylor Model Basin.

II. Plants visited and engineers contacted.

(a) Deutsche Schiffsmaschinen Aktien Gesellschaft (Deschimag) of Bremen - Herr Wiebe.
(b) Fried. Krupp - Germania Werft, Hamburg - Dr. Ing. R. Hoppenrath.
(c) Blohm & Voss - Hamburg - Dr. Kurt Illies and Dipl. Ing. Emil Jarck.

III. Design Details.

(1) **Shafting.**

(a) Materials.

Carbon steel forgings are used exclusively for shafting. The material is in accordance with DIN 35.61 and contains approximately 0.35C. It has a tensile strength of 87,000 PSI (61 Kg/cm^2). Bronze or chromium steel bushings are shrunk on the shafts where these run in water lubricated bearings.

(b) Corrosion protection (See Fig. 1).

Those portions of the propeller and stern tube shafts which are not covered with metal bushings must be protected from salt water corrosion by other means. Shafts were formerly covered with rubber, which was vulcanized to the metallic surface by means of steam passing through the bore of the shaft. During the last year of the war, the rubber coating was replaced by Cellon, which is a plastic cellulose. A coat of Cellon was alternated with a covering of wide hemp ribbon until a thickness of one-quarter inch was attained. An especially strong Cellon coat was applied lastly. A galvanized wire of about 0.2 inch diameter was closely wrapped over the Cellon. To avoid un-winding of the wire as a result of collision with an obstacle, each winding is inter-locked with it-

Design Details, 1(b),(Cont'd).

self. A layer of Cellon is placed over the wire to give a smooth finish.

(c) Shaft Flanges.

These are forged integral with the shaft. The bolt holes are reamed either cylindrical or conical and fitted bolts are used. If a cylindrical bolt is used, a head is provided at one end and a nut at the other end, so that the flanges are held together between the surfaces of the bolt head and nut. The diameter of line shafting is increased at the bearings by about one-quarter of an inch.

(d) Synchronising Coupling.

A toothed type of coupling is supplied between the reduction gear flange and the propeller for disconnecting the shaft in case of a casualty. The coupling has been described in detail in reference (b).

(2) Bearings.

(a) Outboard.

(1) Wood (**Pockholz**) is generally used for bearings operating in water. During the war, phenolic moulded bearings were used as a substitute for wood to some extent. They were found to swell up and grip the shaft, particularly at low speeds. The quality of phenolic moulded bearings was found to be variable.

(2) Roller bearings were tried by Blohm and Voss for shafts running in water. A special emulsifying oil was used for lubricating these bearings, but it was found difficult to protect them from water, and rusting resulted. The roller bearings used were manufactured by "VKF", "Wupperthal Deutschewerft" "OSORNO" (Blohm and Voss) and "HUASCARAN" (Blohm and Voss). The roller bearings had to be replaced after one or two trips.

(3) White metal propeller strut bearings were also in use on a few ships. The bearing was lubricated by a constant head of oil supplied by a tank placed above the water level. Oil entered at the top center of the bearing. The white metal is in accordance with Marine Standard WM 80, and has the following composition:

Sn, 79-81%; Sb, 11-13%; Cu, 5-7%; Pb, 1-3%

(4) Wood bearings are generally loaded to about 28.5 PSI of projected area. This results in a propeller strut bearing having a length of about seven or eight shaft diameters. During the past twelve months, Blohm and Voss constructed wood strut bearings having a length of only one-and-one-half of two shaft diameters.

Design Details, 2(a)(4) (Cont'd).

It was found that these bearings wore less than the longer ones. The also resulted in less hull appendage resistance.

(b) Line Shaft Bearings.

These are lubricated by means of a disc fixed to the shaft and dipping into an oil well. The oil which adheres to the disc is scraped off at the top of the disc and distributed to the top center of the bearing. Ring lubrication is not used due to the danger of breakage and of sticking of the ring to the bearing housing when the latter is not in its normal position.

IV. Design Calculations.

(1) The following calculations are for a geared turbine-driven propulsion shaft of a German cruiser. Maximum speed of the ship is 34.5 knots. Three shafts are supplied, of which the outer two transmit 38750 S.H.P. each at 390 R.P.M. and the middle shaft transmits 14500 S.H.P. at 430 R.P.M.

(2) Calculations for Outboard Propeller Shaft Section I - I, Figure 2.

(a) Area of Section I - I,

$$F_1 = \frac{\pi}{4}\left(\overline{39.5}^2 - \overline{13}^2\right) = 1092 \text{ cm}^2 = 169 \text{ in}^2$$

(b) Polar Section Modulus,

$$W_p = \frac{\pi}{16}\,\frac{(D^4 - d^4)}{D} = \frac{\pi}{16}\,\frac{(39.5^4 - 13^4)}{39.5}$$

$$= 12{,}000 \text{ cm}^3$$

(c) Maximum torque developed,

$$M_d \text{ max.} = 71620 \times \frac{\text{S.h.p.}}{\text{R.p.m.}} = 71620 \times \frac{38750}{390}$$

$$= 7{,}120{,}000 \text{ cm.kg.}$$

Design Calculations (2), (Cont'd).

(d) Maximum Shearing Stress,

$$\tau_{max.} = \frac{Md}{W_p} = \frac{7,120,000}{12000} = 593 \text{ kg./cm}^2$$
$$= 8,420 \text{ p.s.i.}$$

(e) Maximum Propeller Thrust,
Assume propeller efficiency $= \eta_p = 0.64$
Assume thrust deduction $t = 8\%$

$$\text{Propeller thrust, } P_s = \frac{C \text{ SHP } x\ \eta_p}{V_s\ x (1-t)}\ ;\ C = \frac{75}{0.515}$$

$$= \frac{38750 \times 75 \times 0.64}{34.5 \times 0.515\ (1-0.08)}$$

$$= 114,000 \text{ kg.}$$
$$= 253,000 \text{ lbs.}$$

(f) Compressive Stress, Section I - I.

$$\sigma_I = \frac{114,000}{1092} = 104.1 \text{ kg/cm}^2$$
$$1480 \text{ p.s.i.}$$

(g) Bending Stress Due to Propeller Overhang.
Computed weight of propeller, G = 6500 kg = 14400lbs.
Moment Arm = 50 + 10 + 10 (assumed) = 70 cm = 27.5 inches.

The assumed 10 cm. is added to the lever arm since the strut bearing is not rigid.
Bending moment, M_b = 6500 x 70 = 455,000 cm.kg.
= 395,000 in.lbs.

Section Modulus,

$$W_1 = \frac{\pi}{32} \frac{(39.5^4 - 13^4)}{39.5} = 5980 \text{ cm}^3$$

Bending Stress,

$$\sigma = \frac{M_b}{W_1} = \frac{455000}{5980} = 76.1 \text{ kg./cm}^2$$
$$= 1040 \text{ p.s.i.}$$

Design Calculations (2), (Cont'd).

(h) Total Combined Stress, Section I - I.
τ and σ_i are additive once per revolution, so that:

$$\sigma = 76.1 + 104.1 = 180.2 \text{ kg./cm}^2$$
$$= 2560 \text{ p.s.i.}$$

Total combined stress,

$$\sigma_1 = 0.35\sigma + \sqrt{\sigma^2 + 4(\alpha_0 \tau^2)}$$

Where $X_0 = \dfrac{\sigma 2 \text{ allowable}}{\sigma s \text{ allowable}} = \dfrac{800}{(1.3 \times 600)} = 1.025$
(* The origin of the value is unknowns)

$\sigma 2$ = Tensile Strength
σs = Shearing Strength

Substituting,

$$\sigma 1 = -0.35 \times 180.2 + 0.65 \sqrt{(180.2)^2 + 4 (1.025 \times 593)^2}$$

$$= 735 \text{ kg./cm}^2 = 10400 \text{ p.s.i.}$$

(i) Factor of Safety.
Tensile strength of carbon steel forging = 60 kg/mm²
Elastic limit of carbon steel forging = 50 kg/mm²
$F.S. = \dfrac{5000}{735} = 6.8$ based on elastic limit.
$F.S. = \dfrac{6000}{735} = 8.17$ based on tensile strength.

(3) Propeller Shaft, Section II - II, (Fig. 2).

O.D. of Section = 390 mm
I.D. of Section = 255 mm

Polar Section Modulus,

$$W_p = \frac{\pi}{16} \frac{(39^4 - 25.5^4)}{39} = 9500 \text{ cm}^3$$

Shearing Stress, $\tau = \dfrac{7,120,000}{9500} = 750 \text{ kg./cm}^3$

Design Calculations (3), (Cont'd).

Area of Section II - II =

$$F_2 = \frac{\pi}{4}(\overline{39}^2 - \overline{25.5}^2) = 683.9 \text{ cm}^2$$

Compressive stress,

$$\sigma_d = \frac{114000}{683.9} \doteq 167 \text{ Kg./cm}^2$$

Total combined stress,

$$\sigma_i = 0.35 \times 167 + 0.65\sqrt{167^2 + 4\,(1.025 \times 750)^2}$$

$$= 945 \text{ kg./cm}^2 \text{ or } 13400 \text{ p.s.i.}$$

It is to be noted that bending stress is neglected.

Factor of safety $= \frac{5000}{945} = 5.28$ based on elastic limit.

Factor of safety $= \frac{6000}{945} = 6.35$ based on tensile strength.

4. Line Shaft.

This has an O.D. = 380 mm and I.D. = 260 mm.
The bending stress is neglected in computing the total combined stress.
Polar Section Modulus, W_p = 8400 cm^3
Shearing Stress = 848 kg/cm^2
Area of Cross-section = 603.2 cm^2
Compressive stress - due to thrust = 189 kg/cm^2
Total combined stress = 1090 kg/cm^2
F.S. = 4.59 based on elastic limit
F.S. = 5.51 based on tensile strength.

5. Propeller Shaft Coupling, Fig. 3.

The coupling is a carbon steel forging having an elastic limit and a tensile strength of 5000 and 6000 kg./cm^2, respectively. It is acted upon by a torsional moment and the propeller thrust. Both of these tend to open up the coupling at its weakest section, i.e. through the keyway.

Design Calculations, (5) (Cont'd).

The torsional moment is assumed to act through the center of gravity of the cross-sectioned half of the coupling, Figure 3. This has an area of:

$$F = \frac{54 - 39.38}{2} \times 70 = 512 \text{ cm}^2$$

The distance of the center of gravity from the axis is 23.25 cm. Total tangential force,

$$F = \frac{7,120,000}{23.35} = 305,500 \text{ kg.}$$
$$= 673,000 \text{ lbs.}$$

The unit tensile load acting on the cross section, Fig. 3 is:

$$Z_1 = \frac{305,500}{512} = 596 \text{ kg/cm}^2$$

The shaft end has a taper of 1:10, so that the radial thrust is:

$$P_s \times 10 = 1,140,000 \text{ kg.}$$

This force acts on a surface of:

$$F = 67\left(\frac{39 + 32.3}{2} \times \pi - 2 \times 8.0\right)$$
$$= 6430 \text{ cm}^2$$

The unit load is therefore, $\frac{1,140,000}{6430} = 177.5 \text{ kg./cm}^2$

The stress is greatest at the after end of the coupling, at which point it is bored out to a diameter of 39 cm. The stress over a 1 cm. length will be:

$$Z_2 = \frac{39 \times 1 \times 177.5}{2 \times 5.56 \times 1.0} = 622 \text{ kg./cm}^2$$

Total stress is:

$$\sigma_z = Z_1 + Z_2 = 596 + 622 = 1218 \text{ kg./cm}^2$$
$$= 17,400 \text{ p.s.i.}$$

Design Calculations, (5)(Cont'd).

Coupling Factors of safety are:

$$F.S. = \frac{5000}{1218} = 4.11 \text{ based on elastic limit.}$$

$$F.S. = \frac{6000}{1218} = 4.93 \text{ based on tensile strength.}$$

The added strength due to the flange at the forward end and the upset after end of the coupling is neglected.

6. Coupling Flange Bolts.

Material, KM steel, having a tensile strength of 50 kg/mm^2 and an elastic limit of 42 kg/mm^2. Eight (8) conical bolts, with an overage diameter of 95 mm are provided. The thread standard is 77 FG..

The greatest normal load occurs on the bolts when the shaft is operating at full power. This load will be momentarily increased if one of the propellers is fouled and is dragged through the water without turning.

Normal shearing force per bolt:

$$Q = \frac{Md}{8r} = \frac{7,120,000}{8 \times 26} = 34,250 \text{ kg.}$$

Average bolt area,

$$F = \frac{9.5^2 \pi}{4} = 70.88 \text{ cm}^2$$

Maximum shearing stress,

$$\tau = \frac{4 \times 34250}{3 \times 70.88} = 646 \text{ kg/cm}^2$$

A tensile stress, resulting from the initial tightening of bolt and equal to σ_z = 24 kg./cm^2 must be combined with the shearing stress.

The combined stress is:

$$\sigma_i = 0.35\,\sigma_z + 0.65\sqrt{\sigma_z^{\,2} + 4\,(1.025\,\tau)^2}$$

$$= 0.35 \times 24 \quad 0.65\sqrt{24^2 + 4\,(1.025 \times 646)^2}$$

$$= 868 \text{ kg/cm}^2$$

Resulting factors of safety are:

$$F.S. = \frac{5000}{868} = 5.78 \text{ based on tensile strength.}$$

Design Calculations, (6)(Cont'd).

$$F.S. = \frac{4200}{868} = 4.85$$ based on elastic limit.

7. Key for Outboard Coupling, (Figure 4.)

Number provided = 2, 180° apart.
Size = 80 x 40 mm

For the portion of the key in the shaft, the tangential load is:

$$\frac{7,120,000}{2 \times 16} = 222,200 \text{ kg.}$$

The load carrying surface of each key is:

$$(2.1 - 0.3)(66 - 2 \times 2.5) = 109.8 \text{ cm}^2$$

The unit pressure between the key and seat is:

$$\frac{222,200}{109.8} = 2024 \text{ kg/cm}^2$$

$$= 28,800 \text{ p.s.i.}$$

A similar calculation shows the bearing load of the key in the coupling to be 1850 kg/cm² or 26,500 p.s.i.

8. Calculations for Middle Shaft.

The calculations of ref. (a) are for the shafting of a three (3) shaft cruiser which develops the following powers on the shafts:

	Ahead		Astern	
	HP	RPM	HP	RPM
Port and Starboard shafts	38750	390	13500	236
Middle Shaft	14500	430	14500	328

Ahead calculations for the middle shaft are made in the same manner as for the outboard shafts.

9. Strength of Middle Shaft with Astern Operation at 14500 SHP.

(a) Revolutions of propeller for astern operation.

For this condition, the outboard shafts transmit 13500 S.H.P. so that the total power for astern operation is:

Design Calculations (9)(Cont'd).

$$2 \times 13500 + 14500 = 41{,}500 \text{ S.H.P.}$$

Assuming 35% slip, and an estimated speed of 20 knots, the R.P.M. of the middle propeller is obtained from:

$$V = \frac{n\ H\ 60\ (1-S)}{1852}$$

Where V = speed in knots
n = RPM
H = defined by Fig. 5 = 2.9 meters.
S = slip, = 0.35

Substituting

$$20 = \frac{n \times 2.9 \times 60 \times (0.65)}{1852}$$

$$n = 328 \text{ RPM.}$$

(b) Pull of Propeller.

The following formula is used to comput the pull of the propeller on the shaft for astern operation:

$$P_a = \frac{(s.h.p.)_a\ 75 \times \eta_p}{V \times 0.515\ (1 - t)}$$

where

η_p = propeller efficiency for astern operation = 40%
V = astern speed in knots.
t = thrust deduction = 0
$(s.h.p.)_a$ = astern s.h.p.

$$P_a = \frac{14500 \times 75 \times 0.40}{20 \times 0.515\ (1-0)} = 42200 \text{ kg.}$$

(c) The mechanics involved in computing the stressed in the shafts, bolts, flanges, etc., for astern operation is similar in other respects to that used for ahead operation.

10. Strength of a locked outboard shaft at a speed of 14 knots. (Fig. 5).

(a) Pull developed by a locked propeller.

Design Calculations,(10)(Cont'd).

A coupling is provided in each of the outboard shafts for disconnecting the propeller from the turbine at 14 knots, or 7.2 m/sec, in case of damage to one of the outboard shafts or to the propulsion machinery. In this case, the remaining outboard shaft and the middle shaft are considered to be driving the ship at 14 knots and the locked outboard propeller develops a torque which is a function of this speed and its characteristics. The pull developed by the propeller is computed from:

$$N = \psi \gamma A_a \frac{V_m^2 \cdot \sin^2 \epsilon}{2g} \text{ (Kg.)}$$

where

ψ = 1.45
γ = 1025 (weight of $1m^3$ of sea water)
A_a = developed area of propeller - $9.0m^2$
ϵ = angle between tangent to propeller helix at 0.7 R and shaft centerline (see Fig. 5).
g = 9.81 m/sec.^{2}
V_m = 7.2m/sec at 14 knots.
$\tan \epsilon = \frac{0.7 \pi D}{H} = \frac{0.7 \pi \times 3.46}{3.74} = 2.03344$

$\epsilon = 63^\circ \; 48' \; 43.2''$

$\cos \epsilon = 0.44128$

$\sin \epsilon = 0.89737$

substituting:

$$N = 1.45 \times 1025 \times 9.0 \frac{(7.2)^2(0.89737)^2}{2 \times 9.81}$$

N = 28450 kg. pull developed by the dragging outboard propeller.

(b) Torque developed by locked propeller is given by the equation:

$$= 0.7 \times \frac{D}{2} \cdot N \cdot \text{Cos}$$

where

D = defined by figure 5, centimeters
N = pull developed by propeller.
ϵ = angle between tangent to propeller helix at 0.7R and shaft centerline.

Design Calculations (10)(Cont'd).

Substituting,

$$M_d = .7 \times \frac{346}{2} \times 28450 \times 0.44128 = 1{,}525{,}000 \text{ cm.kg.}$$

(c) Stress in Shaft Due to Locked Propeller at 14 knots.

The total combined stress, due to a pull of 28450 kg and a torque of 1,525,000 cm.kg., is computed in the usual manner and found to be 217.6 kg/cm^2. This is much less than for normal power ahead operation of the outboard shafts.

(d) Strength of an Outboard Shaft when Uncoupling at Full Power Ahead.

The total horsepower developed by the three shafts at full power is 92000 s.h.p., corresponding to a speed of 34.5 knots. In case of a casualty to one of the outboard shafts the ship will continue for a short time at full speed then gradually slow down to a speed corresponding to the reduced horsepower. From the view-point of high shaft loading, it is assumed that the ship still has a speed of 34.5 knots for a brief period after the damaged shaft has been stopped.

The pull exerted by the propeller for this case is:

$$N = 1.45 \times 1025 \times A_a \frac{V_m^2 \sin^2 \epsilon}{2}$$

Where

$A_a = 9.0 m^2$ developed area

$V_m = 17.8$ m/sec

$\sin \epsilon = 0.89737$

$g = 9.81$ m/sec^2

Substituting,

$N = 173{,}900$ kg. pull on shaft.

The locked torque developed by the fixed shaft is (see Fig.5).

$$M_d = 0.7 \times \frac{346}{2} \times 173{,}900 \times 0.44128$$

$$= 9{,}300{,}000 \text{cm.kg.}$$

The value of propeller pull and locked torques are used as above to obtain total combined stress in the shaft.

11. Safety of Shafting Against Buckling.

The thought has been advanced that there is a possibility of the longest shaft buckling under propeller thrust load at full power,

Design Calculations (11)(Cont'd).

Euler's long column formula applies for this case,

$$K = \frac{2\pi^2 E J}{\ell^2}$$

Where

J = bending moment of inertia

$$= \frac{\pi}{64}(\overline{38}^4 - \overline{26}^4) = 80{,}000 \text{ cm}^4$$

E = modulus of elasticity for carbon steel
2,150,000 kg/cm^2
3,057 x 10^6 psi

ℓ = length of shaft from forward end of stern tube to thrust bearing

1580 cm for port outboard shaft.

Substituting,

$$K = \frac{2 \times 9.87 \times 2{,}150{,}000 \times 80{,}000}{(1580)^2}$$

$$= 1{,}365{,}000 \text{ kg.}$$

Since the propeller thrust is 114,000 kg, the factor of safety of the port outboard shaft against buckling is:

$$F.S. = \frac{1{,}365{,}000}{114{,}000} = 12$$

For the starboard outboard shaft, it was necessary to provide line shaft bearings with cast steel covers to raise the factor of safety against buckling to 8.7.

Prepared by:

R. Michel
Technician.

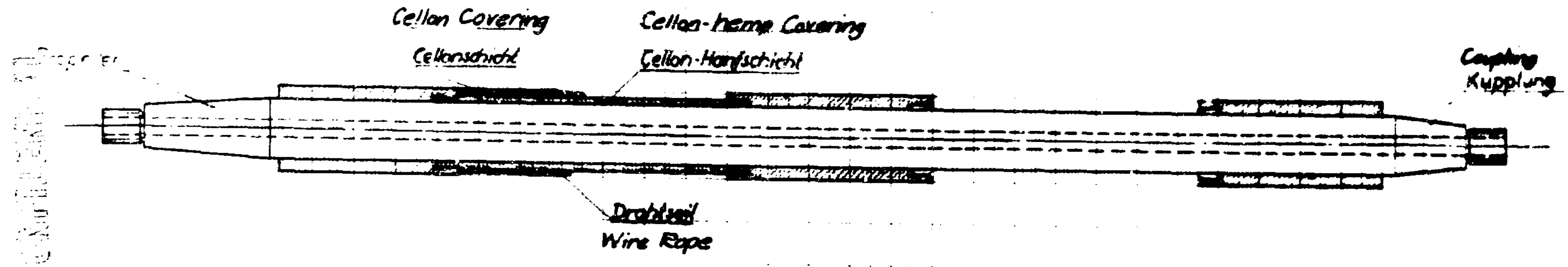


FIG. 1
CORROSION PROTECTION
FOR
PROPELLER SHAFTS

23/8/45

A. Für [illegible] Vorwärtsfahrt mit 3 [illegible]

I. Schraubenwellenfestigkeit [illegible] schnitt I e 1

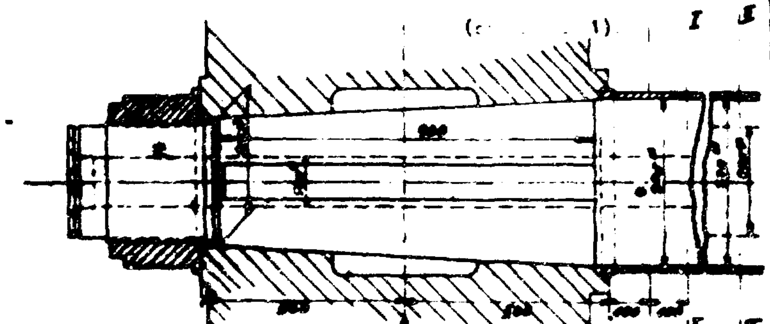


Fig: 2

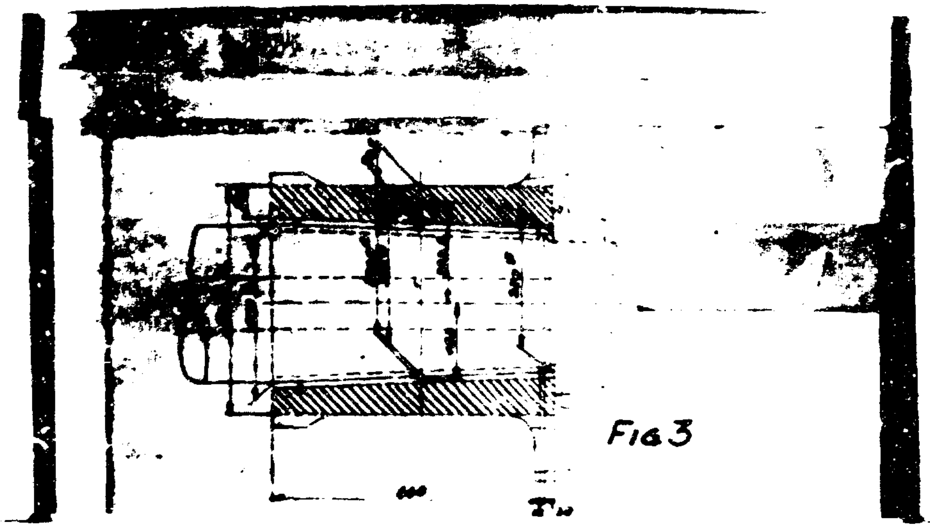


FIG 3

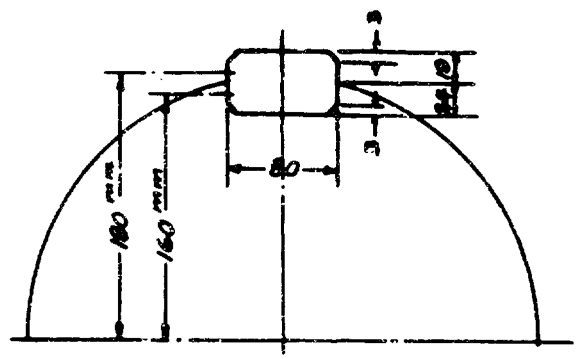


Fig. 4

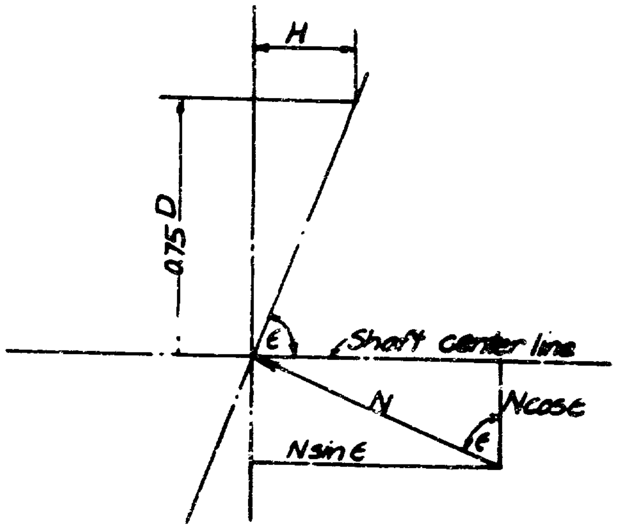


Fig. 5

CONFIDENTIAL

APPENDIX

Kiel - Gaarden

20 May 1945.

Field of Study of Dr. Ing. R. Hoppenrath.

Friedrich Krupp Germaniawerft A.G., Kiel - Gaarden.

1. Strength Investigations.

(a) Theoretical investigations of the strength relationships in foundations for motors, turbines, etc.

(b) Strength measurements of machine parts, foundations, and pressure hull parts.

(c) Measurements and tests of materials.

2. Vibration Investigations.

(a) Theoretical vibration calculations of mass-elastic systems for surface ships and submarines (Torsional vibrations).

(b) Theoretical calculations of vibration relationships in foundations for main and auxiliary machinery.

(c) Measurements of torsional vibrations in shafting of surface ships and submarines - Adjustment of vibration dampers for Diesel motors.

(d) Shock measurements in foundations.

3. Acoustics.

(a) Theoretical pre-calculation of sound dampers for the exhaust for Diesel motors for surface ships and submarines.

(b) Calculations and consultation regarding noise reduction measures for elimination of the spread of noise in the ship. The acoustic treatment was applied to air spaces as well as the hull.

(c) Most suitable form of steel foundations in order to reduce transmission of hull vibration as much as possible.

(d) Determination of the most suitable sizes of resonance damping devices of rubber and steel springs.

(e) Measurements of air resonance conditions in surface and underwater ships.

Appendix (Cont'd).

(f) Measurements of hull resonance conditions in surface and underwater ships.

(g) Continual checks on correct installation of air and hull resonance damping devices during production of submarines.

4. Mechanics of Shock.

(a) Theoretical research in the dispersion of shock in the hull and foundations of surface and underwater ships.

(b) Measurements made on board surface and underwater ships to determine the characteristic amplitudes of the shock process resulting from the detonation of mines and depth charges.

(c) Experimental research of the ability of main and auxiliary machinery and other devices on board surface and underwater ships to withstand shock.

(d) Experimental determination of the most suitable dimensions of foundations, mountings, screwed or threaded joints or connections, and other fastenings to resist shock loads.

(e) Determination of methods for reproducing the shock curves measured on ship board during the detonation of depth charges and aerial bombs.

---

1. (a) "Theoretical Research in Strength Relationships."

The ordinary strength computations, based on the familar formulas of "Bach" and "Thum", were made by the various construction departments themselves. More difficult problems, where stress and strain relationships could not be clearly seen, were solved at the request of the individual departments. This was largely a matter of strength and deformation tests and research on the foundations of machinery and ships which involved statically indeterminate problems. Projects usually followed the methods of higher theoretical mechanics. For instance, the mathematical computation of the deformation in a double buttom was made by considering the bottom as a grid of girders. The deformations at the inter-sections of the longitudinal and transverse supports were determined with the aid of influence lines (graphical solution).

Appendix (Cont'd).

The constants for the necessary equations were taken from the values of the elastic deflection lines. Thus one can arrive at the actual spatial deformation of the entire double bottom. One may then take into consideration the effect of moments at the reactions at internal and outboard bulkheads. The stresses in built-up foundations then are computed from the deformations thus determined.

1. (b) Strength Measurements.

In order to determine the best relationship between the structural parts of machinery, foundations, etc., special experimental set-ups were made which were then strength tested in the laboratory. In these tests all the probable forces arising during operation were simulated. The measurements of strains were made with the Berg, Thum, Lehr, and Huggenberg instruments. Gage lengths of 20, 10, 3 and 1.5 mm were used. The last mentioned gage lengths were used especially for determing the stress peaks in transition points of the structural parts. Enlarged models were often used in the case of smaller objects in order to determine their elastic curves. Stress and strain tests were made on actual surface and underwater craft to get an idea of the deformation and load relationships on board.

1. (c) Strength tests and Measurements of Materials.

In order to keep a check on production, test samples were taken from the individual structural parts, which were then tested for their physical properties in the laboratory. These tests had to do with determining the hardness and tensile strength; bending tests, and notched bar impact tests were made in special cases dynamic strength tests and measurements were made. For still further classification of materials, photomicrographs of the material were sometimes used. In addition, the individual composition of the material was in many cases determined by chemical analysis.

2. (a) Theoretical Pre-Calculations of Torsional Vibration Relationships.

In order to eliminate the overstressing of the shafts of surface and underwater ships from torsional vibrations, the range of natural frequencies and the amplitude of the vibrations were mathematically computed in advance. The natural frequencies were

Appendix (Cont'd).

determined either by the Tolle or Holzer methods. The masses of the driving as well as the driven machines or propellers were introduced in the form of the so-called "mass moment of inertia" - - cmkgsec^2. In this the elasticity of the shafts was used in the form of the stiffness constant:

$$l_e = \frac{l_w}{G \cdot I_p}$$

In this, l_w = the equivalent length in cm. of the shaft section being studied; G = the torsional modulus in kg/cm^2; I_p = the polar moment of inertia of the cross-section in cm^4. If several shaft sections of different diameter and different material are present, the elastic constants - called elastic lengths - must be computed separately for the various sections. Total or overall elasticity is the sum of the individual elasticities. The mass moments of inertia are computed by the formula $\Theta = m. \; r^2$ in cmkgsec^2. In the case of Diesel motors, the actual variability of the reciprocating masses was considered, such as the piston and the connecting rod, by computing the mean value. That is, the purely rotational masses were inserted in the last mentioned formula in their full values or magnitudes. The reciprocating oscillating masses, on the other hand, were only assigned 50% of their true magnitude. The method of computing was to introduce the approximate value into the tabular computation and then to investigate the system for balance using the approximate value of the natural frequency. According to Holzer's or Tolle's suggestion, the exact natural frequencies can then be determined for the individual assumed frequency values from the curve of balanced moments. In order to obtain the magnitude of the forced natural stresses plus those caused by resonance vibration, it is useful to analyze harmonically the diagrams of the pressure curves in the motor cylinder. This is done by drawing or computing the tangential pressure diagram that goes with the indicator card and then determining the harmonic components by one of the familiar methods such as "Hummel": "Harmonic Analysis". In multi-cylinder engines of 6 cylinders and up the torsional vibration relationships in the cam shafts were also pre-determined, and the dimensions of the shafts were set according to the vibration conditions prevailing. The dimensions of the shaft were so fixed, if possible, that natural frequencies did not occur in the usual speed ranges. In multi-cylinder engines, such as are

Appendix (Cont'd).

almost exclusively used today, this is not usually possible, however, since the higher harmonic components follow one another too closely. Hence, it is necessary to use so-called vibration dampers. The Sandner system was mainly employed. The basic prinicple of the system is that rotating masses are increased or reduced hydraulically by means of the acceleration forces resulting from torsional vibrations . This increase is done automatically by a valve controlled by springs. Such dampers are usually placed at the points of greatest vibrational amplitude. Such a point is the free end of the crankshaft, and it is here that the vibration damper is usually attached. The control fluid is taken directly from the hollow crankshaft in the form of lubricating oil. The virtual increase or decrease of the masses results in a displacement of the natural frequency by 15 to 20%. The automatic control valve in the vibration damper is usually adjusted to a pressure of between 12 and 15 atm. In computing the stresses in advance, only the propeller and engine damping are considered. Damping resulting from friction within the bearings is not included since its influence is neglible. The damping of the propeller is decidedly important for the first frequency of the system. It is computed according to "Holzer". For the first natural frequency the damping effect of the motor can be disregarded. In the case of higher natural frequencies, only the motor damping exerts any appreciable influence on the amount of stress. The damping itself is obtained from the following equation based on experience:

$$q = \frac{D^2 \cdot H^2}{1000}$$

D is the cylinder diameter in cm, and H is the piston stroke in cm. Then, the damping moment is $M_D = q \cdot \omega = q \cdot 2\pi \cdot n$. If the stressed due to torsional vibration exceed the value of $\tau = 300$ kg/cm^2, a vibration damper must be used, since otherwise a break in the shaft may occur. In order to still further reduce the danger of a break in the shaft, care must be exercised to see that all points of shaft discontinuity are rounded off to as large a radius as possible. If this is not done, high stress concentrations will occur at such points. This greatly reduces the permissable load limit (by about a factor of 2). In determining the natural frequency of propeller shafts with propellers, one must observe that the moment of inertia of the propeller is essentially increased by the water entrained with it. Experience has shown that this increase amounts to about 25 - 30%.

Appendix (Cont'd).

2. (b) Theoretical Pre-calculation of Vibration Relationships in Foundations.

Any natural frequency computations presuppose a knowledge of the elasticity and mass relationships in the vibrating foundations, etc. In the majority of cases any exact determination of the effective characteristic amplitudes is impossible. However, in order to be able to make predictions about the probable behavior of motors, turbines, gears, etc., on their foundations, it is assumed that the dynamic flexure line coincides with the static one. One determines the static flexure line by one of the familiar methods and then in further calculations takes into account the effect of the masses of the supports, foundations, etc. Then the real natural frequency can be approximated accurately enough by one of the approximation methods, such as that of Dunkerly, Timoschenko, or others. In computing the rigidity of foundations one must consider if the girder is riveted or welded. In riveted girders the effective moment of inertia is essentially smaller than the geometric moment. In the case of welded foundations with large plate surfaces, some reduction in the geometric moment of inertia must also be made. Damping moments at the ends of the girders and additional supports must also be considered in computation. At any rate, the calculation must be made for the girder with the fixed ends and once also for the girder with freely supported ends. It is useful to solve these problems graphically, since in mathematical computation the solution soon becomes difficult to grasp, while in graphical solution one usually has some check on the method of solution.

2. (c) Measurement of Torsional Vibration.

On the basis of experience the natural frequencies can nowadays be determined accurately enough. Nevertheless, it is useful to make a check measurement. This has to do first with determining the range of natural frequency and then with determining the greatest amplitude of torsional vibration. If a layout has just been built for the first time, in other words, if it is a new type layout, then measurements can be made synchronously at several points. The form of the vibration can then be checked from the individually determined amplitudes. This and the range of natural frequency give a sufficiently clear picture of the range of the stress actually present in the elastic systems. In every case the mathematically computed vibration form and the twist of the shaft section given with it are used as the basis for determing the stress caused by torsional vibrations. If the vibrations are too great, it is advisable to readjust the vibration damper that may be already installed. This can be done by regulating the pressure valve in the type of vibration damper used here, the Sandner type. The torsional vibrograph built by "Deutsche Versuchsanstahlt der Luftwaffe" as well as the Geiger Torsiograph are used for taking the measurements. As a running check on vibration dampers, an electric indicating (inductive) torsional vibration indicator was attached to the primary part of the Sandner dampers.

2. (d) Measuring Vibrations Transmitted to Foundations.

Sometimes natural frequencies are set up in the foundations of auxiliary machinery and similar devices by the forces of the driving motors themselves. Thus, alternating stresses can occur which may lead to the cracking or breaking of foundations, mounting bolts, or other securing members. Since it is impossible in the majority of cases to calculate the natural frequencies in advance, the exact range of the natural frequency can only be determined by actual measurement with the Vibrograph. The Geiger vibrograph was used for this purpose. The contact vibrometer built by Askania was also used because of its simple operation. If the natural frequency lies within the normal operating speed range, it must be reduced by suitable methods. This may be accomplished by changing the masses or by increasing or reducing the rigidity of the foundation. It has been shown that for structural and operational reasons there is only one practical change in the rigidity of the foundation that makes any effective difference in the range of natural frequencies. If it is at all possible, one should try to increase the rigidity, that is, to shift the frequency upward. At low pitch, that is, at low natural frequencies, one must reckon with a very strong transmission of the vibration energy to the whole ship. Local natural frequencies join with the induced vibrational movements to cause very unpleasant beat phenomena (out-of-tune beats). This must be looked out for especially in ships having more than one propeller shaft. People have tried to eliminate the beat phenomena at low pitch by using synchronizing devices. This has not been completely successful for many reasons. If one has not observed these relationships while planning the ship, or if some late change in the ship's structure has resulted in such beat or coincidence phenomena, then it is better to rectify the condition by changing the speed of one of the engines. Elastic intermediate members are sometimes used in order to lower the frequency of main and auxiliary machines. These may be made either of rubber or in the shape of steel springs. There has been a shift in recent times to a steel spring mounting because of the inability of rubber to support loads. It is always necessary to observe during assembly whether the unit is one that produced free forces, or whether it is one that is completely balanced. In the first case the six possible natural frequencies around the three spatial axes must be mathematically computed in advance. By selecting the mounting or bearing points or by changing the stiffness of the springs one can so control the frequencies that they don't lie within the normal operating range. If the normal operating range is too large, that is, if the changeable operating speed extends over several hundred rpm, then one is forced to choose a mounting that is either super or sub-critical. That is, in the first case, one tunes the whole layout to so low a frequency that all the natural frequencies lie sufficiently far below the lowest limits induced. But here one must be careful that the stability of the whole unit be not reduced by such a low frequency. In rough seas, in pitching and rolling, the engine exerts a great thrust against the rigid foundation or against the hull. Hence, it is simpler to choose the second kind of mounting, in which case one doesn't have the above mentioned difficulties to deal with. But one must accept a reduction in the damping effectiveness.

3. (a-g) Research in Acoustics.

Sound spreads throughout the ship in part through the resonance of the air and in part through the resonance of the hull. Special attention must be devoted to the propagation of sound through piping or other ducts, such as, for example, the exhaust pipes of Diesel motors. The natural vibration relationships can be computed in advance by the well known methods: of. The Theory of Sound, by Lord Raleight or Acoustics by Steward and Lindsay. In ducts that have branches or that are otherwise complicated, it has proven useful to employ approximate methods with the help of gradual approximation for determining the natural frequencies. The

Appendix (Cont'd).

mathematical method is fundamentally the same as is used in determining the natural frequencies of mechanical systems. In order to reduce the spread of noise, sound or noise dampers (noise filters) are used. In surface ships there is usually enough room to install the very useful low flow filter (Tiefpassfilter). In the very limited space conditions on submarines it is useful to employ the high flow filter. Since a considerable part of the noise energy is transmitted through the foundations one can exert a considerable influence on the amount of hull resonance energy transmitted by properly designing the steel foundations. In calculating these relationships in advance it is again a question of determining the natural flexure frequencies and the natural longitudinal frequencies. While the method of determining the latter is treated in detail in the above mentioned work on vibration technology, there is only a very little about figuring the natural flexure frequencies in composite systems. Any calculation is complicated by the fact that four marginal conditions must be met. Hence, it is more practical not to compute but to study the matter of the transmission of sound through the hull by flexure frequencies from a model. In this instance measurements made on full size objects produce the quickest results. Hull resonance is measured by means of a so-called hull-resonance feeler. In it the movement of the surfaces is changed by induction into corresponding electrical values. The movement processes of hull resonance can thus be made directly perceptible to the ear either through earphones or loudspeakers. In making objective measurements it is nevertheless desirable to fix the electric values by means of a galvanometer. The transmission of hull resonance or hull noise plays an important role in the audibility of submarines. Hence, not only were continual experiments being made on full scale objects, but production was carefully checked in this respect.

4. (a-3) Mechanics of Shock.

In order to reduce as much as possible the shock stresses on the hulls and installations of surface and underwater ships caused by the detonation of depth charges or bombs, people began to develop special constructions. Since the forces that occur during such underwater detonations are so great that one cannot re-enforce or stiffen construction enough to withstand them, it is necessary to change to a type of construction having the greatest possible flexibility or ability to shift without to much deformation or stress.

Appendix (Cont'd).

Thorough tests were made on full scale objects in order to determine the magnitude of the forces. Measurements were made by attaching accelerometers to the outer hull wall or to the pressure hull at the points where greatest stress would occur. These accelerometers consisted of quartz pressure elements, in order to obtain as high a frequency as possible -- about 25 kg cycles. The movement processes during detonation occur during an extraordinarily short period of time - some where between 0.1 and 1/1000 sec. The dependence of the greatest amplitudes of acceleration on the distance of the detonation point from the ship's outer hull or pressure hull was determined. This function is shown on Illustration Nr. 302 for various measuring points. Velocity and distance measurements were made similarly. For instance, Nr. 301 shows the dependence of shifing caused by the shock of depth charges. Nr. 308 shows the time curve of related values of course velocity and acceleration for one measuring point on the pressure hull. Nr. 303 shows the reduction of the acceleration peaks dependent upon the distance of the detonation for three different measuring points.

Pressure measurements were made in the water near the outer hull along with the acceleration measurements. At 40 m detonation distance, for example, a maximum pressure of $p = 140$ kg/cm^2 was determined. This is, of course, for the primary shock wave. This pressure wave is reflected at the outer hull or at the pressure hull. Some of the shock energy passes over into the pressure hull when the latter is deformed. Another part is reflected. In the German boats tested, about 1/3 of the shock energy in the water passed over into the pressure hull in the form of deformation energy. Nr. 298 shows the time curve and the dependence of the explosive charge and the detonation distance for some basic measurements.

Along with these measurements strain measurements were also made on the inner walls of the pressure hull or on the ribs. These were generally made tangentially around the periphery of the hull or in the direction of the boat's longitudinal axis. Nr. 304 shows some results of these measurements. The greatest stresses occur longitudinally, both as tensions and compressions. The greatest stresses were recorded locally in the plate surface, greater than at any other points. Nr. 304 shows the stress curve for the individual measuring points as a function of the detonation distance. From this it may be seen that at a detonation distance of about 21 m the greatest stresses, with an amplitude of

Appendix (Cont'd).

about 45 kg/mm^2 occured, along the longitudinal axis of the plate surface. (See curve 20b). Around the periphy, however, there was only a pressure of about 10 kg/mm^2. (See curve 20a).

The bending or distortion of the entire pressure hull was measured in the case of another boat with partly riveted hull. The detonation tests were carried up to the point where the boat was actually destroyed. Measurements were made with a telescopic measuring device which showed the time curve (See Illustration 299) as well as the locational curve for individual measuring points that were distributed longitudinally over the entire boat. (See Nr. 299 and 300).

Nr. 306 shows some examples of shock absorbing installations of auxiliary machinery etc., and the time curve of the probable shifts that might occur. It may be seen from the measurements that the shock process can be conceived of as a vibration process. In contrast to the usual calculations of vibration processes, it is, rather, a matter of determining the process at the very start of vibration. As a manner of approaching the problem, one can also figure from the stationary condition. Then one can make use of the normal resonance curves for determining the damping values. This damping value shows the relationship or ratio in which the shock amplitudes are blocked as they pass from the outer hull, through various elastic intermediate members, into the mass of the machinery unit concerned. Nr. 305 gives an example of determining the damping values for the most commonly occuring shock initiating frequencies. Here the probable damping figure is entered over the natural frequency of the elastic mount (lower Figure 2), while the initiating frequencies contained in the shock curve introduced as parameters. This results in an average damping number in the magnitude of 1 : 850. Therefore, in the example selected, a primary acceleration at the outer pressure hull was reduced by elastic intermediate members between the machine and the pressure hull to a pitch of 20 cycles, or by approximately @ 1/850.-g (g = acceleration of gravity) there is an acceleration of about 10 g at the machine mass itself. The force can then be determined from the machine mass and the acceleration.

It will be seen from this law that the low frequency impulses are more dangerous than the high frequency primary accelerations.

Propeller hum.

In surface and underwater ships there occurs a humming of the propeller at certain speeds. This phenomenon is so pronounced as to be audible throughout the entire ship. It occurs not only in German ships, but also in non-German ships, as may be easily shown by references to the literature on the subject.

This phenomenon is a disadvantage in passenger ships because the hum makes it almost impossible to stay in the after compartments of the ship. In the case of submarine vessels this phenomenon is undesirable, especially at low speeds, because it becomes impossible to procede noiselessly, as in tracking. It is important to note that the phenomenon occurs chiefly at low speed ranges.

All sorts of tests have been made to try to learn what causes this humming. As far as is now known, no effective means of removing this humming have been found. At any rate, no published works are known that contain suggestions for such reduction. However, Dr. Gutsche, in the magazine "Werft, Reederei und Hafen", published an essay on the results of experiments with propellers which were aimed at getting some clear proposals for doing away with propeller hum. From his point of view, the humming, which means a vibrating of the propeller blades, may be traced back to a breaking up of eddies as the inciter of the vibration phenomena. According to these studies, special variations from the streamlined form of the propeller blade cross sections are responsible for inciting the vibrations. He computes the probable frequency from measurements on the profile errors.

According to this, the starting of the vibrations depends essentially on the flow conditions around the propeller profiles. Hence, it would be supposed that with increasing rpm or with increasing peripheral velocity and hence, with increasing velocity of flow, the exciting of vibrations would also increase. It has been determined, on the contrary, that an increase of the volume, that is, of the amplitude of the vibrations, can be observed at low rpm ranges. The curve of resonance or sound volume does not increase or fall monotonously over the whole range of rpm, but characteristic peaks show up. This condition suggests that the question is one of the propeller itself. On the other hand, it has often been supposed in various publications, that it is a question of resonance phenomena within the ship's hull near the propeller. This should be caused by eddy currents near the propeller. The fact remains, that it may be observed that within the rpm range where humming occurs more less strong vibration or oscillation phenomena may be recorded.

CONFIDENTIAL

Appendix (Cont'd).

The nature of the humming that has been observed, however, makes it seem unlikely that the plates of the hull could be responsible for the resonance phenomena. It can be observed that the humming is quite pure, that is, that it is a matter of the generation of an almost pure tone with only a few overtones. Hence, it may be deduced that whatever vibrating parts are concerned are not made up of many structural members. It must, rather, be a question of some homogenous, almost completely self-contained body. Hence, the hull of the ship, with its many breaks and additional stiffening supports can hardly be considered as producing such a pure kind of hum.

In order to get some kind of clear picture of all these matters, extensive tests and measurements were made here. The measurements were made exclusively on submarine craft. It had been suspected that the struts and bearings near the propeller exerted an influence on the humming. There was also the possibility that the humming could have been indicated by the flood ports near the propeller, or that the separating strips between the flood ports might have a tendency to vibrate, or hum. In order, to get at this question the rudder control linkage, the net guards, and a part of the rudder were dis-assembled, one after the other, and a measuring run was made each time without these parts. In order, to determine their influence, the flood ports near the propeller were closed. The propellers themselves were normal propellers such as were in common use on all Germans boats up to that time. These measures proved to have no influence on the generation of a hum. In the rpm ranges between $n = 60$, the lowest possible number of rpm, and the highest rpm that was of interest $n = 300$ min^{-1} the humming occured in the definite ranges in the same form with the same frequency and with the same entensity. Thereupon it was suspected that the propeller cowling which covered the nut holding the propeller on the hub end of the shaft and the shaft sleeve where the shaft emerges from the after shaft bearing might have an influence on the propeller hum. Again, test runs were made with the various parts dis-assembled in stages. These also produced negative results.

This proved then, that the humming or singing was caused exclusively by the propeller. Hence, the following experiments only dealt with shaping the propeller or exchanging it. In order to have comparable objective results, the attempt was made to measure the time curve as well as the intensity of the sound that

Appendix (Cont'd).

was being radiated. At first, the group listening device of the boat itself was used as a measuring device. (G.H.G. - Layout). It was soon seen that because of the great distance separating them, the interference caused by the flow phenomena along the boat prevented any useful or accurate re-production of the measurements. Certain details of the physical process at the origin of the noise were especially difficult to understand. Hence, it was necessary to move the pick-up apparatus as close as possible to the propeller itself. A Seignette quartz receiver was used as receiver for the noise frequencies, similar to the one that is built-in stationary in the hull as part of the G.H.G. layout. In order, to eliminated the disturbing flow noises at the pick-up itself, caused by the breaking up of eddies at the edges of the quartz, it was necessary to streamline the pick-up itself. The pick-up was suspended by means of a tube of sufficient length and strength. The tube itself was so fastened to the hull with collars and braces, that various areas near the propeller could be covered during surface or underwater movement. The quartz itself and the leads had to be made waterproof. A cathode-ray oscillograph with two recording points was used as a viewing or recording apparatus. In order, to keep the frequencies as pure as possible an electric filter was cut in between the transmitter (the quartz) and the registering device. (Band pass filter). The viewing apparatus was in the boat itself.

In order to follow the radiation of noise from the pressure hull toward the inside of the boat at the same time, hull resonance pick-ups were used near the propeller. In this pick-up the movement amplitudes of the skin of the pressure hull were converted by induction into corresponding electric processes. With the help of a calibrated amplifier whose frequency characteristic runs linearly throughout almost the entire audible range, the electric voltages were amplified so that they could be either read off in a calibrated galvanometer or so that the voltages could be registered as to their time curve in the above mentioned cathode-ray oscillograph. Revolving drums or continuous film feed were used for the actual recording. In each case care had to be exercised that the time solution of the procedure be large enough to make details of the sound process recognizable.

These measurments, both those with the Seignetter quartz and those with the hull noise pick-ups had the same qualitive results. Because, the measuring process was so much simpler only the hull noise pick-ups were used in the later experiments.

CONFIDENTIAL

Appendix (Cont'd).

As the subjective observations had already shown, the sound process during the humming revealed a decided vibrational character. The main frequency range in which the humming appeared was between 400 - 800 cycles. This same frequency range appeared in both surface and underwater ships. The only difference between these two types was that the rpm range in which the humming occured was greater in the case of underwater travel than in surface travel. The amplitudes are also greater in underwater travel than in surface travel. In spite of the same general construction, individual propeller revealed differences in their tendency to hum. If both port and starboard propellers showed the same tendency, it was immaterial whether one went with the port of the starboard propeller. The humming could be stopped by choosing definite differences in the rpm of the port and starboard propellers. This was the first means used to avoid humming or singing within a limited range, but it was not always successful.

In order to find out what influence propeller material had upon the origin of humming, propellers were made of different materials. The original common bronze propellers were replaced by ones made of cast iron or cast steel. Experiments were also made with a chrome nickel steel alloy of the Krupp works at Essen, called "P 125". It was found that the material had nothing to do with the origin of the singing. Only case iron showed any influence due to its natural damping effect, since in its case the range in which the humming occured was smaller than with the usual materials.

Since it was suspected that the form of the propeller blade had some essential influence on the origin of the humming, bead-like reinforcements were put around the outer edge of the propeller balde in a tangential direction to the direction of flow. The shape of these beadings was varied in all sorts of ways, but this change of shape proved to have no measureable influence on the originating of the humming.

A solution of the problem was first obtained by making the profiles of the individual propeller blades different among themselves. These differences only need be very slight. Greater differences in the profiles result in terrific forces within the propellers, which do eliminate the humming, but which cause additional noises from the propeller shafts pounding in their bearings. Hence, the differences in the profiles must only amount to a few millimeters. Such changes are unrecognizable.

CONFIDENTIAL

Appendix (Cont'd).

Another proposal for effectively eliminating the humming is to sharpen the streamlined entering edges of the propeller blades like chisels. The change in the profile must be such that there must be a pronounced edge at the place where the sharpened part joins the rest of the propeller surface.

Use of both of these proposals resulted in a sure elimination of propeller humming or singing. These measures were also advantageous since they could be carried out later on propellers that had already been complted or that were mounted. All German submarines were equipped with such altered propellers, based on the principles proposed by the Germaniawerft.

These measures, then, influenced the tendency to vibrate. Thus it seemed logical to damp the propeller blades themselves. The following is one effective way to damp propeller blades: Slots were sawed perpendicular (at right angles) to the periphery at the points of greatest vibration amplitudes in the propeller blades. These points are chiefly on the out circumference of the blades. The slots were anything up to about 100 mm. long. The edges of these slots are then bent together in such a way that they touch all along their entire length and form a rubbing closure. If one equips at least one propeller blade with this kind of slots, its inner damping is increased so that humming no longer occurs, or, so that amplitude has become so slight that it can no longer radiate noise energies to its surroundings. In order to avoid rubbing, which actually amounts to additional strain and an increase in inner stresses, it was proposed to use instead of the friction slot a cut about 20 mm wide and about 100 mm deep at right angles to the periphery of the propeller and to insert some kind of friction body in this cut.

The friction body must be so constructed that it can't be forced out of the slot by centrifugal force or by the various forces of flow around the propeller. To that end it is made in the form of a double cone. The inclination of the oblique surfaces is only about 10 - 15°, but must be large enough so that there is no blocking between the material of the propeller and the friction body itself. The body is also to be inserted with several hundredths of a millimeter play. A test submarine was equipped with propellers of such design, and no singing or humming could be detected. Nevertheless, because of the greater danger from pressures as a result of the slot cut in the blades any further application of this method of damping was given up.

Appendix (Cont'd).

Influence of bearings on propeller hum.

In making measurements with the hull noise pick-up it was noted that pronounced frequencies were present near the after shaft bearing. The frequencies extended essentially over the lower and middle range of audibility. Beside the frequency between $\nu = 400-800$ cycles there occured pronounced amplitudes between $\nu = 100-300$ cycles. The frequencies in the first mentioned ranges were more pronounced with propellers that hummed than they were with propellers that did not hum. Any measurement of the phases was impossible because of the attendant difficulties and the inaccessiblity of the waves to be measured. Hence, the direct connection between propeller humming and generation of noise in the bearings could not be clearly shown. It was also observed that the pulsation amplitudes continued down into the sub-sonic range (mechanical pulsations and oscillations).

From the observations made, it was possible to determine that some relationship existed between the generation of noise in the bearings and the propeller hum. Since it was technically impossible to measure this and thus prove it, and since there was a growing scarcity of the (Pockholz) Lignum Vitae almost exclusively used as a bearing material in this kind of bearing, it was hoped this might be changed to some domestic kind of material. Hence, the proposal was made to replace it with either hard or soft rubber rods. The rubber was to be a buna product. In those instances where plastics and other synthetics had already been tried as bearing materials there had been considerable trouble. This was because it had been impossible to find a synthetic that would not swell. The material Novotext, which swelled very little, had to be replaced with Novozell. The first of these was a synthetic made of plastic with a textile filler; the second was a molded plastic with cellulose fiber filler. The first material caused a lot of trouble by swelling, but the second one swelled so much that in many cases there were breakdowns in the drive system caused by freezing of the shaft bearings. It is interesting to note in this connection that the bearings of synthetic materials had about the same noise generating qualities as those with the usual wood rods. It might also be mentioned that in each case the bearings were only lubricated with sea water.

Because of the completely different surface qualities of buna rubber as compared with the pockwood or synthetics and because of the completely different surface elasticity relationships, it was expected that the bearing noise might be changed by the switch

Appendix (Cont'd).

to buna rubber. Hence, at our proposal about 20 submarines were equipped with stern bearing whose bearing rods were of buna.

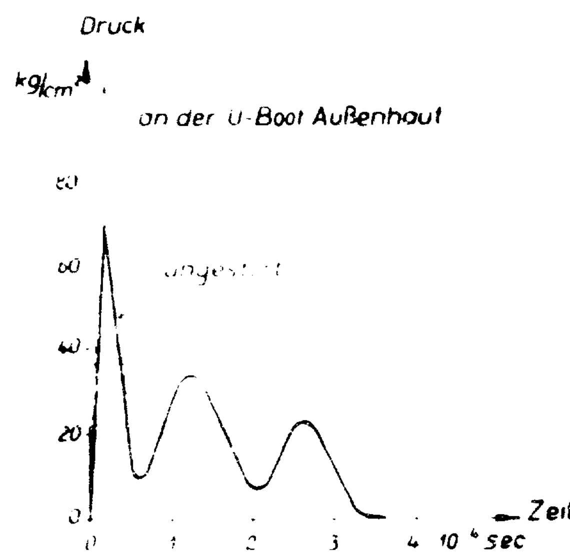
In the first boat a set of normal propellers were used. The entering edges of the propeller blades were not sharpened into chisel shape nor were the propeller profiles differentiated. Normal propeller bronze was also used as the propeller material.

The measurements made during the test run of this craft showed that the driving speed ranges between $n = 60 - 300/min^{-1}$ the propeller hum had been entirely eliminated. A repitition of the test with runs made at greater depths gave the same results. But instead of humming, within the normal ranges, another disadvantage showed up. When the engines were turned off, that is while the shafts were slowing down from their slowest drive speed of about 60 rpm to zero there was a brief but powerful dissonance in the frequency range between $\nu = 300$-800 cycles. This conditions was repeated on all the other boats. It is interesting to note that this dissonance occured when the engines were turned off in spite of changing the propellers in the manners described above. For this reason the rubber rods were later replaced by those of Lignum Vitae in the stern bearings.

This experiment showed that propeller hum can be not only a feature of flow around the propeller itself, but that there must also be a direct connection between noise phenomena at the propeller and the noise phenomena at the bearings. In order, to follow this out, the Krupp Germaniawerft designed and built a special testing device with which these relationships might be more closely studied. The test layout was completed and was only used slightly up to the present writing (May 1945). The brief experiments conducted only served to strengthen the above views. Circumstance prevented further experiments. It would, however, be of special interest to use this test layout for studying the lubrication phenomena in such bearings. The results of such study would be of use not only for submarines but also for surface ships.

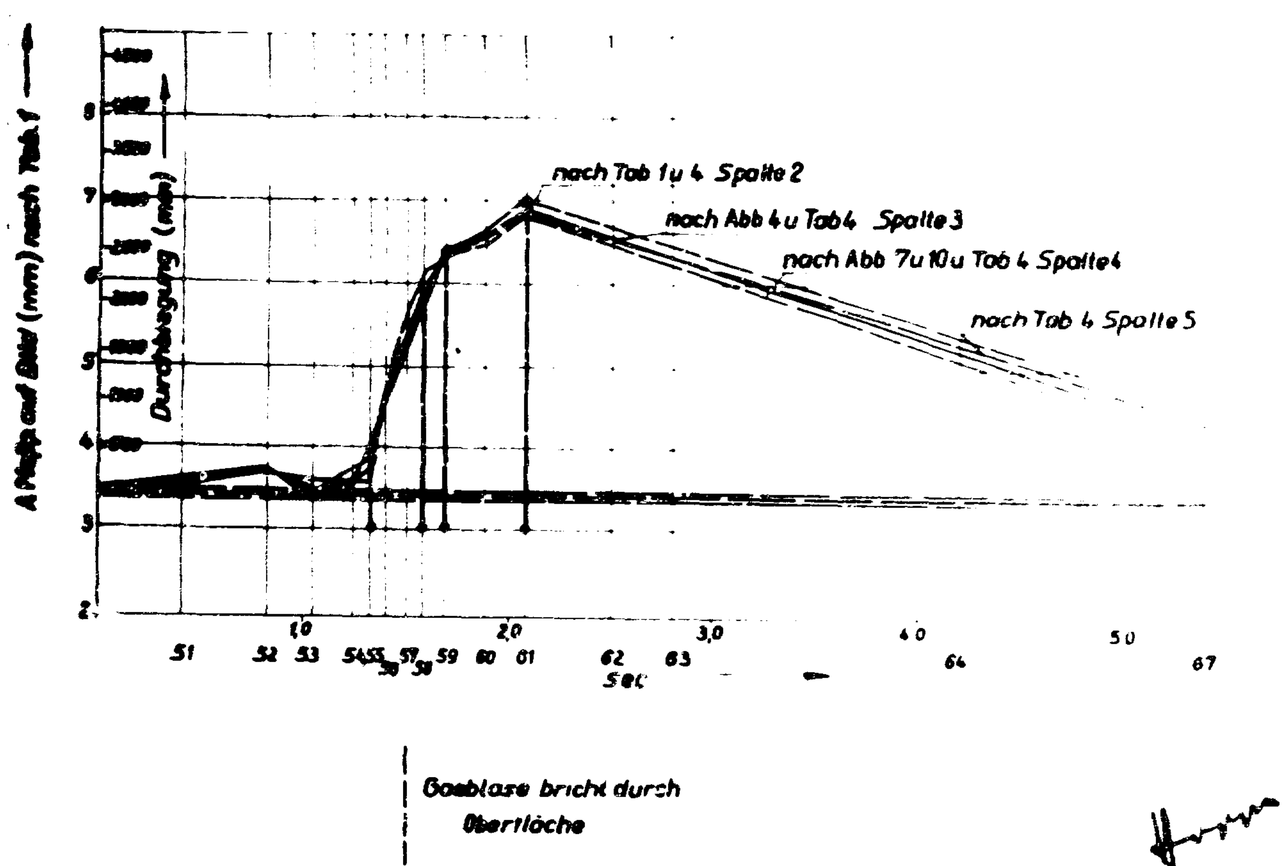
Druckverlauf

im Wasser

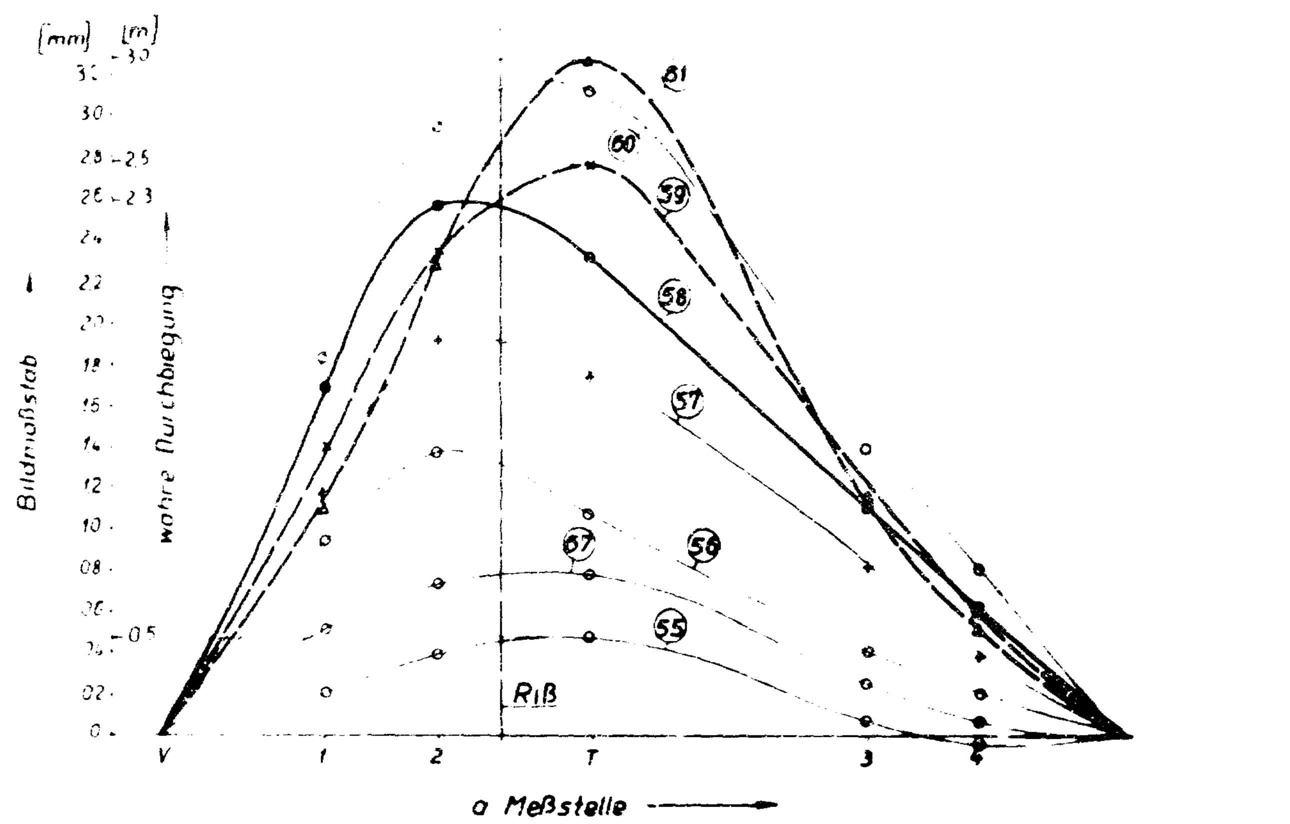


Ladung kg	Abstand m	P_{max} kg/cm²	P_r kg/cm²	Δt Millisec	Entfernung a cm		$\frac{P_r}{P_{max}}$
1	6	110	36	0.43	35	146 10⁴	0.33
1	6	80	31	0.46	33	144 10⁴	0,39
125	40	73	30	0.30	22	146 10⁴	0,41
125	40	72	33	0.28	20	142 10⁴	0,46
125	40	140	60	0.43	31	144 10⁴	0.43

40



Krupp	Zeitlicher Verlauf der Durchbiegung von „US 1"	WpA Nr 299



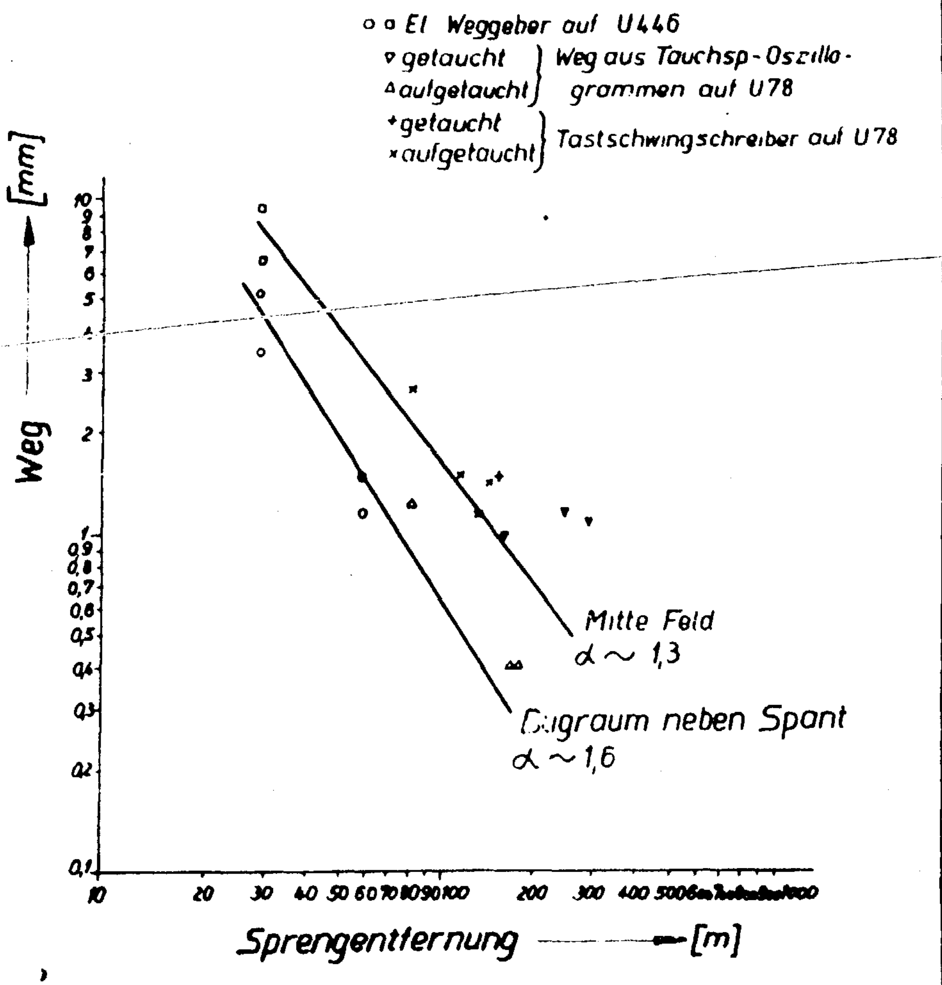
Fried. Krupp Germaniawerft Werkstoffprüfstelle	**Durchbiegung des Bootes „US1"** bei einer WBD Sprengung 18,5m unter Kiel	WpA Nr. 300

CONFIDENTIAL

CONFIDENTIAL

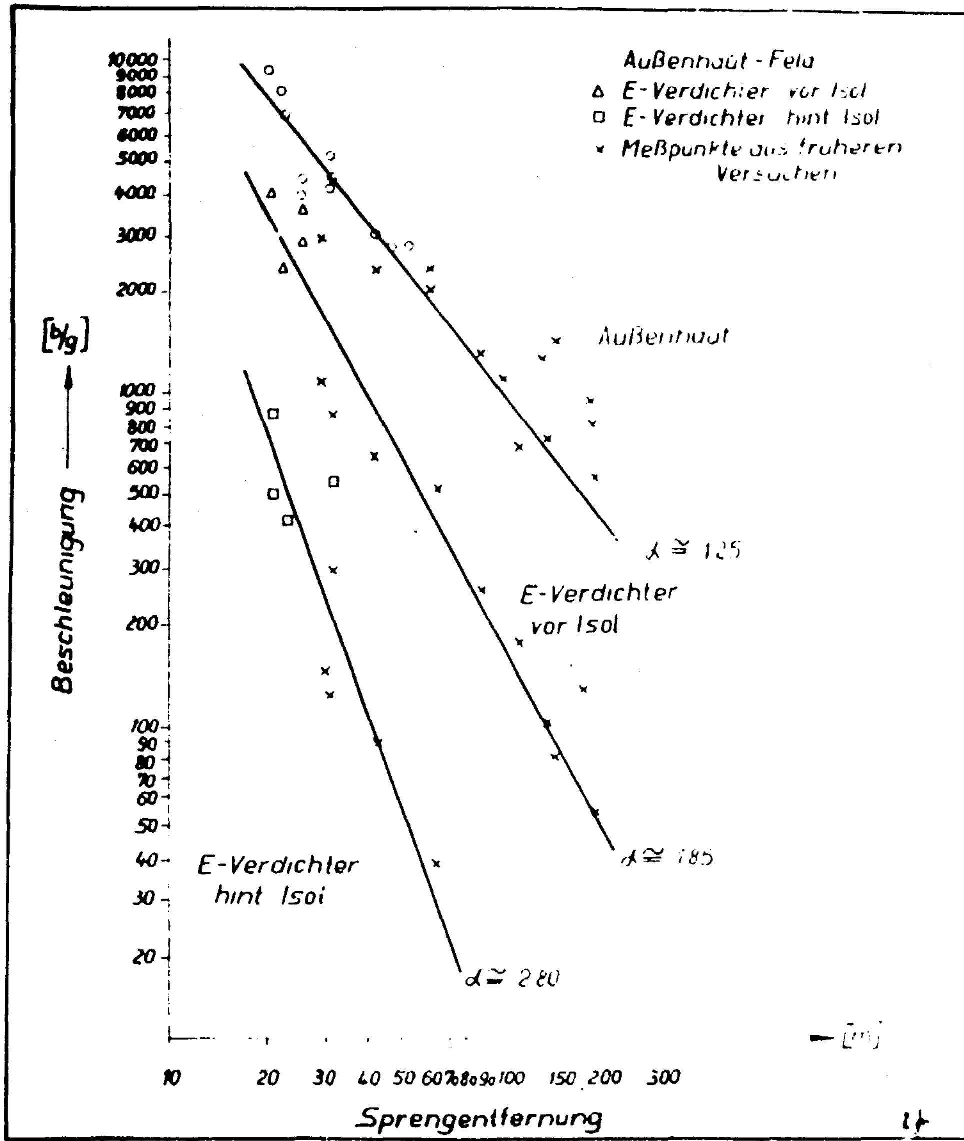
WpA Nr. 301

Maximaler Weg an der Außenhaut von U-Booten (Typ VII C) bei Ansprengungen mit Wasserbomben (WBD) (CPVA)



Fried Krupp
Germaniawerft
Werkstoffprüfstelle

CONFIDENTIAL



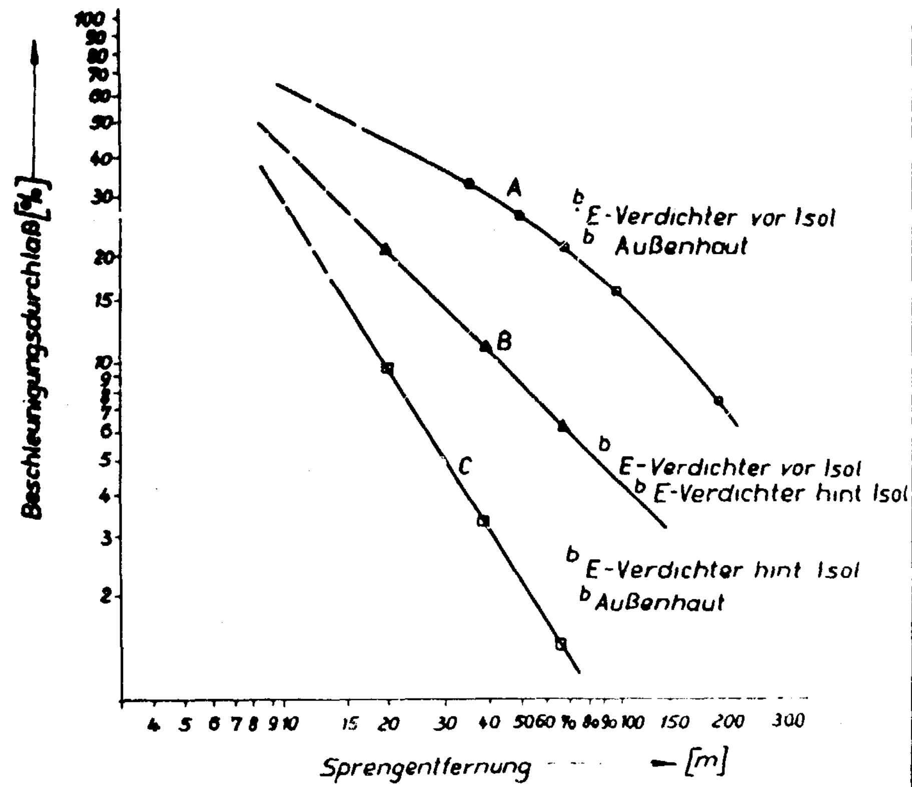
W.A. Nr 302

Beschleunigung als Funktion der Sprengentfernung

Krupp
Werkstoffprüfstelle

Wpa

Nr 303



Beschleunigungsverhältnis $\frac{\text{E-Verdichter}}{\text{Außenhaut}}$ als Funktion der Sprengentfernung

Fried Krupp
Germaniawerft
Werkstoffprüfstelle

Nr. 304 - MISSING.

BuShips Code 654 (Technician R. MICHEL) is being requested by letter to locate copies believed available there and forward to all activities receiving this report.

KpA

Nr 30…

Stoßberechnung für Zersetzer

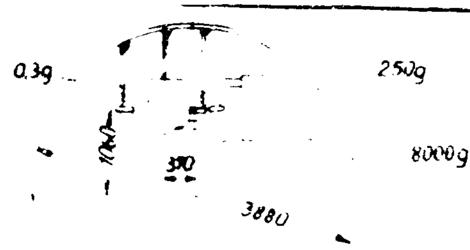


Fig. 1

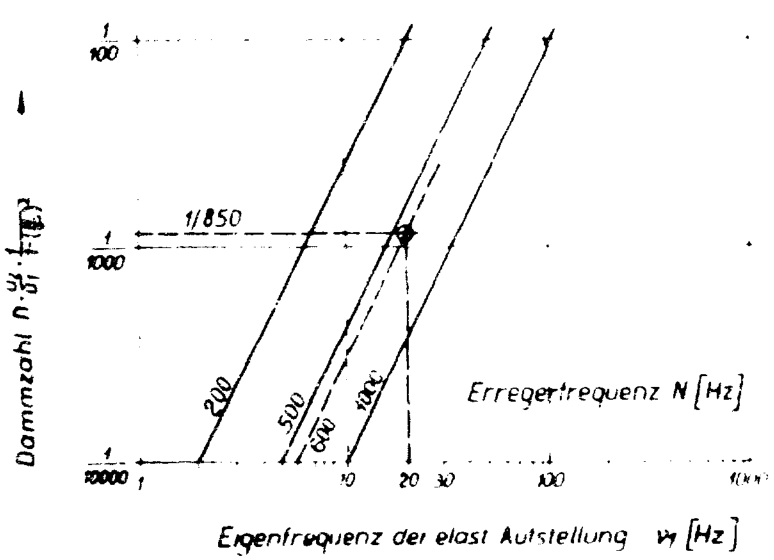
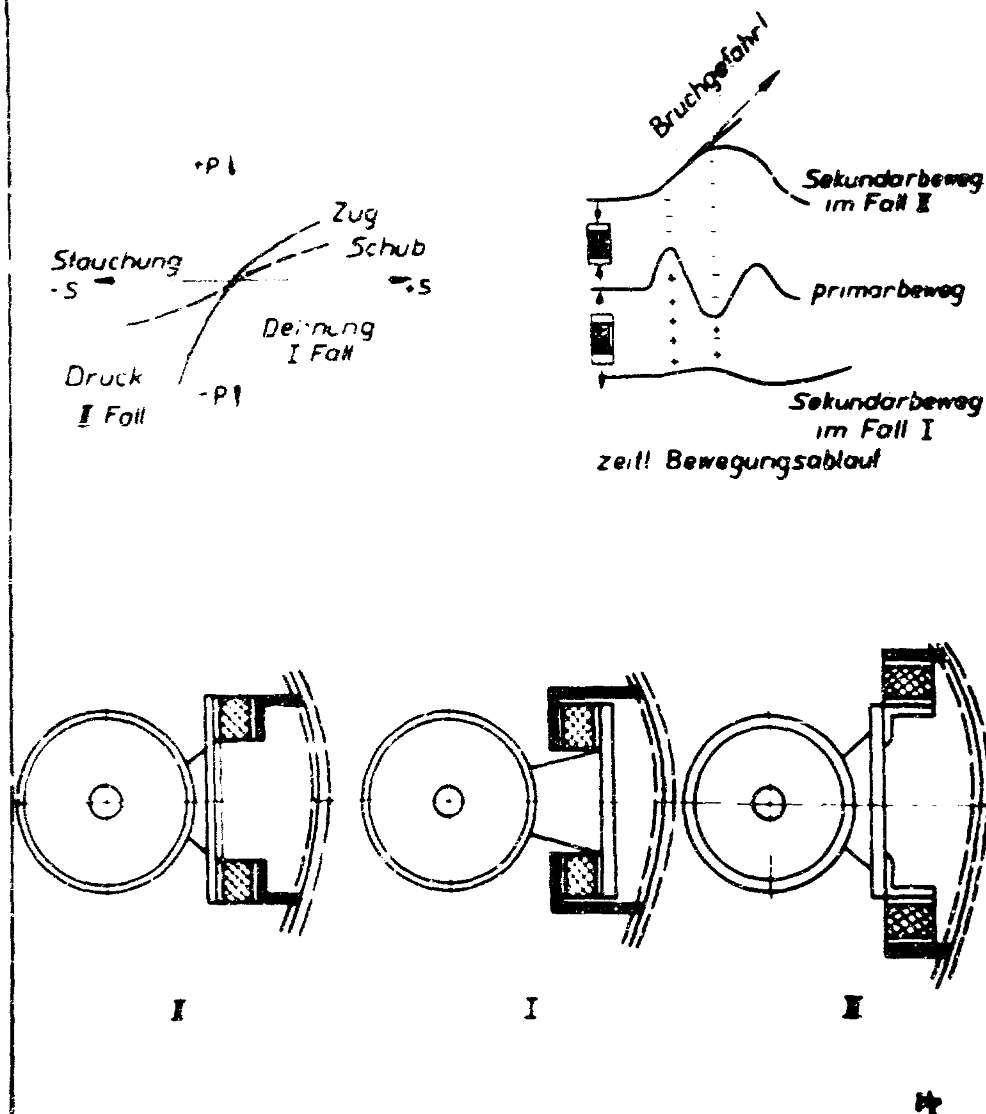


Fig. 2

Fried Krupp
Germaniawerft
Werkstoffprüfstelle



WpA

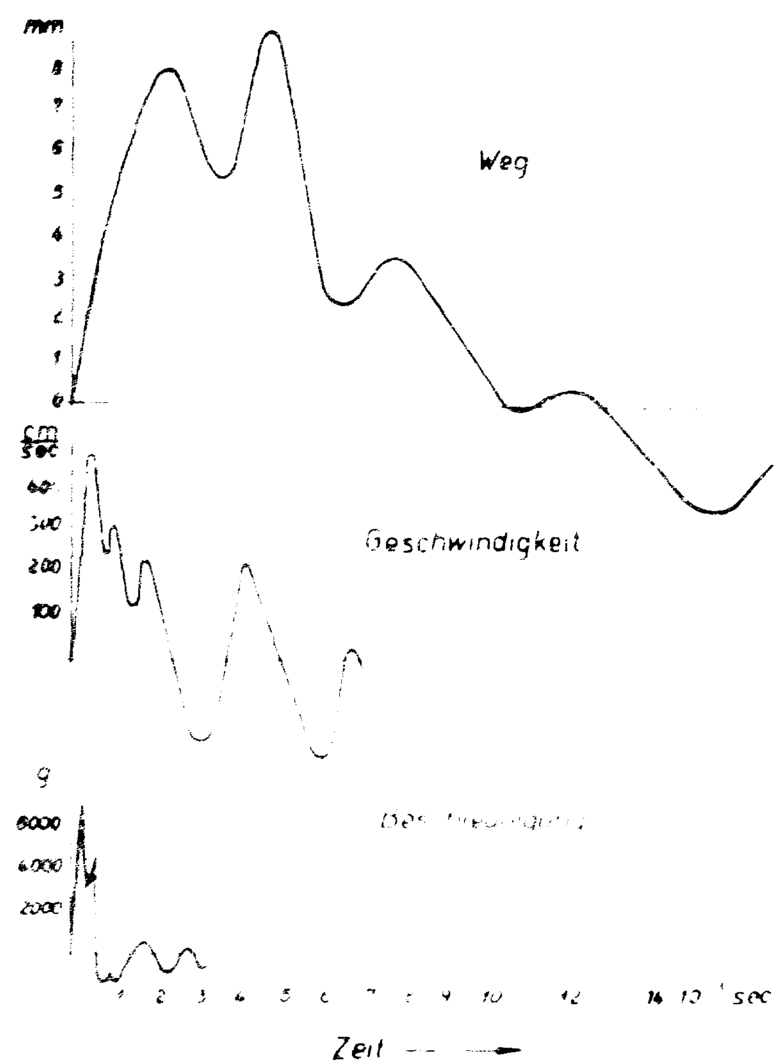
Nr 308

Schema der Stoß-wirkung auf versch. elast. Aufstellungen

Fried Krupp
Germaniawerft
Werkstattprüfstelle

CONFIDENTIAL

WpA Nr 308



Weg-, Geschwindigkeits-u. Beschleunigungsverlauf an der Ubootsaußenhaut bei Detonation einer WBD in 28 m Abstand (CPVA)

Fried Krupp
Germaniawerft
Werkstoffprüfstelle